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STORAGE RELIABILITY OF MISSILE MATERIEL PROGRAM

RAYTHEON

RAYTHEON COMPANY
EQUIPMENT DIVISION

LIFE CYCLE ANALYSIS DEPARTMENT
HUNTSVILLE, ALABAMA

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MISSILE HYDRAULIC AND PNEUMATIC SYSTEMS ACTUATOR ANALYSIS

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10 ~~Prepared by:~~ Joe C. Mitchell

Approved by: Donald R. Earles

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ABSTRACT

This report documents findings on the non-operating reliability of hydraulic and pneumatic actuators. Long term non-operating data has been analyzed and storage failure rate predictions have been made.

This report is a result of a program whose objective is the development of non-operating (storage) reliability prediction and assurance techniques for missile materiel. The analysis results will be used by U. S. Army personnel and contractors in evaluating current missile programs and in the design of future missile systems.

The storage reliability research program consists of a country wide data survey and collection effort, accelerated testing, special test programs and development of a non-operating reliability data bank at the U. S. Army Missile Command, Redstone Arsenal, Alabama. The Army plans a coninuing effort to maintain the data bank and analysis reports.

This report is one of several to be issued on hydraulic and pneumatic devices and other missile materiel. For more information contact:

Commander
U. S. Army Missile Command
ATTN: DRSMI-QSD, Mr. C. R. Provence
Building 4500
Redstone Arsenal, AL 35809
Autovon 746-3235
or (205) 876-3235

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SECTION 1

INTRODUCTION

Materiel in the Army inventory must be designed, manufactured and packaged to withstand long periods of storage and "launch ready" non-activated or dormant time. In addition to the stress of temperature soaks and aging, they must often endure the abuse of frequent transportation and handling and the climatic extremes of the forward area battle field environment. These requirements generate the need for special design, manufacturing and packaging product assurance data and procedures. The U. S. Army Missile Command has initiated a research program to provide the needed data and procedures.

This report covers findings from the research program on hydraulic and pneumatic actuators. The program approach on these devices has included literature and user surveys, data bank analyses, data collection from various military systems and special testing programs.

A failure rate prediction has been derived from the storage time data and failure mode and mechanism knowledge.

SECTION 2

SUMMARY

Over 864 million part hours of storage data were collected and analyzed. This represents data from seven missile programs, three space programs and searches of literature and data centers.

Identification of the part types on the data analyzed was limited to hydraulic or pneumatic types. Failure rates for these types were estimated as shown in Figure 2-1. The numbers in parenthesis show the range of failure rates observed for programs showing at least one failure. The upper 90% confidence limits are 269 fits for hydraulic and 118 fits for pneumatic actuators.

FIGURE 2-1. ACTUATOR FAILURE RATE SUMMARY

ACTUATOR TYPE	NON OPERATING FAILURE RATE (FITS)*	OPERATING FAILURE RATE (FITS)*
Hydraulic	(9.8) 199 (40769)	15,228
Pneumatic	(63.) 88 (256)	1507

*Fits - Failures per billion hours

Also shown are the corresponding operating failure rates in a ground environment. This data was taken from the RADC Nonelectronic Reliability Notebook.

SECTION 3

PART DESCRIPTION

Actuators discussed are those used primarily in missile systems. Actuator classification for the purpose of data analysis and collection is structured as shown in appropriate military standards and publications.

The part descriptions of various types are detailed in respect to those most common to missile applications. Other types are described briefly. Those most common hydraulic actuators are the hydraulic, linear, rotary, piston, plunger and vane type. Pneumatic types are control and springless diaphragm. Motor controlled consist of electric, electro-pneumatic, electrohydraulic, gas motor, and combinations. The solenoid actuator is basically a pilot-operated type.

3.1 Hydraulic Actuators

Missile thrust vector control systems that utilize jet vanes in the rocket-engine exhaust nozzle, movable nozzles, or aerodynamic vanes for thrust vector control require the application of a force or torque for control. In hydraulic thrust vector control systems, this force or torque is supplied by a hydraulic actuator through a suitable linkage.

There are two fundamental types of hydraulic actuators -- linear and rotary. The linear type is more commonly used for this application. It consists of a tube with a highly finished bore in which a piston head and rod reciprocate. A cap at either end, with fluid ports, completes the basic construction. Seals are provided at the piston head and piston rod. (Two other types of construction not so commonly used are the bellows and the diaphragm configurations; these will not be discussed further, since no applications of this type were found during the investigation.) Rotary actuators are designed with a limited rotary motion output, usually less than

360 degrees; they are often called oscillating motors. Hydraulic motors provide a continuous rotary output. Rotary actuators can transmit motion directly, thereby eliminating linkages associated with linear actuators. However, linear actuators appear to predominate for the thrust vector control application -- possibly because of their relative simplicity of construction and the years of experience gained in their use in aircraft applications.

3.1.1 Piston actuators (rams or jacks) are made either with double piston rods or with a differential piston. The piston rod passes through end covers without any seals. This is permissible, because this particular ram is located below oil level. The pistons have no seals; they are accurately machined and lapped, permissible clearances being from 0.03 to 0.04 mm. A cut-away view is shown in Figure 3-1.

In order to minimize wear between the piston (and its seal) and the cylinder walls, the cylinder walls must be hard and smooth. Cylinder barrels are made of heat-treated alloy steel (or sometimes stainless steel) or of high-strength aluminum alloy. Several problems have arizen with respect to galvanic and/or stress corrosion in hydraulic cylinders. Proper selection of cylinder, piston, and piston-rod materials will minimize galvanic action. The entry of water in the oil accelerates galvanic action, so the entry of moisture into hydraulic systems through breathers and other means should be prevented. Stress-corrosion cracking has been observed by one manufacturer in cylinders made of precipitation-hardened stainless steel (17-4PH) and precipitation-hardened aluminum alloy (7070-T6). The mechanism of stress-corrosion cracking occurs in metals which contain residual stresses from the forming process and which are attacked by a corrosive medium. Cracks result if the localized surface notches formed by the corrosive attack cause the residual stress to exceed the strength of the material. Since the corrosive attack usually creates notches at grain

boundaries, resulting in intergranular failure. Precipitation-hardening heat treatments often create a susceptibility to this phenomenon, because during such treatment the precipitation of a second phase occurs first at the grain boundaries. This initial precipitate is noncoherent and does not contribute to hardening, but may accelerate the corrosive attack at the grain boundaries. The coherent precipitate that provides the hardening effect occurs on preferred crystallographic planes within the crystal. Whether stress-corrosion cracking occurs depends to a great extent on the precipation-hardening treatment used. One manufacturer reports that, with 17-4PH stainless steel, stress-corrosion cracking occurred when heat treatment H900 was used, but did not occur when H925, H1050, or H1100 was used. A micrograph of the 7079-T6 aluminum-alloy cylinder showed that the crack was definitely intergranular. The probability of stress-corrosion cracking can be minimized as follows:

1. Design the cylinder forging so that the material in the center containing high residual tensile stress is machined away.
2. Anneal the part following the forging or other forming method.
3. Use a heat treatment that does not accelerate intergranular corrosion; a solution heat treatment is preferred if the necessary hardening can be obtained.
4. Minimize the corrosiveness of the medium contacting the metal; the chlorine ion is especially destructive to chromium steels.

3.1.2 <u>Plunger actuator</u>

Plunger hydraulic rams (Figure 3-2) are the cheapest and the most convenient to manufacture, because they require accurate machining only along the short length of cylinder bore taking the bush. The first type of plunger actuators were provided with double O-seals and subsequent types with double O-rings.

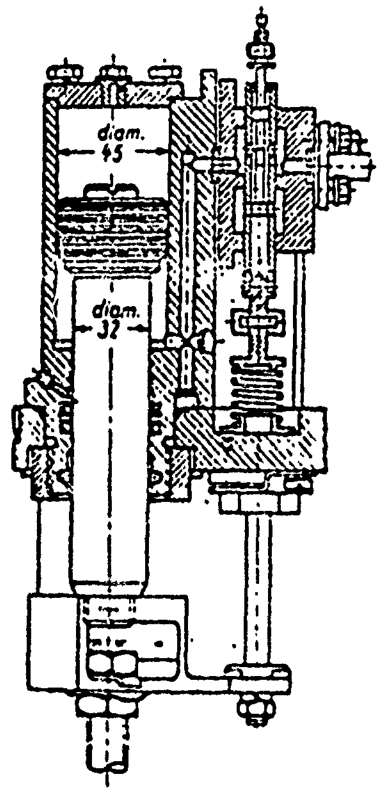

Figure 3-1. Piston type Hydraulic actuator (ram) with a differential piston.

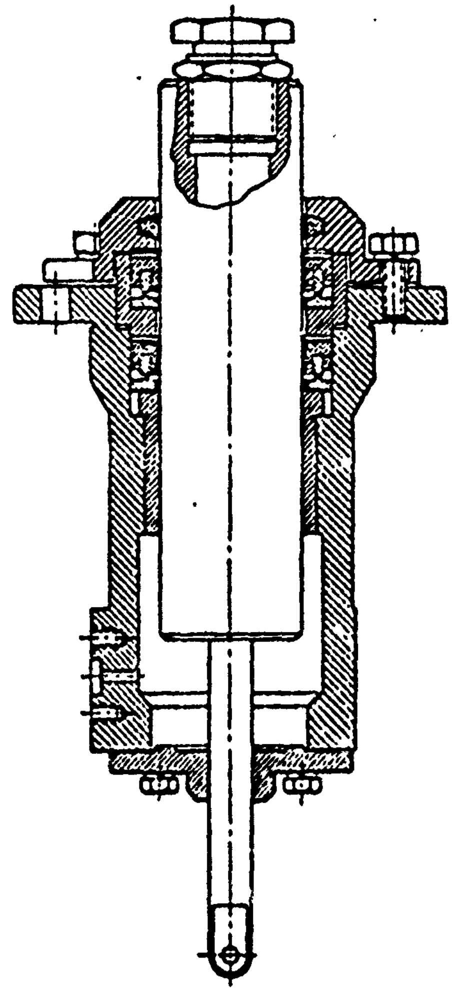

Figure 3-2. Plunger type hydraulic actuator (ram).

3.1.3 Vane actuator

The vane type hydraulic actuator (abutment) is very convenient for reciprocating angular motion.

A peculiar feature of this hydraulic actuator is the hydraulic dashpots, which cushion the impact of the vane against the stop at the extreme of movement with sharp reversal. A cut-away view is shown in Figure 3-3.

3.2 Combination Actuators: Electropneumatic, Electrohydraulic, Pneumohydraulic

Use of a servo-valve for precise positioning of pistons is a well established technique. A servo system is a closed control loop within itself. The system requires one of the standard command signals which may be electrical or pneumatic.

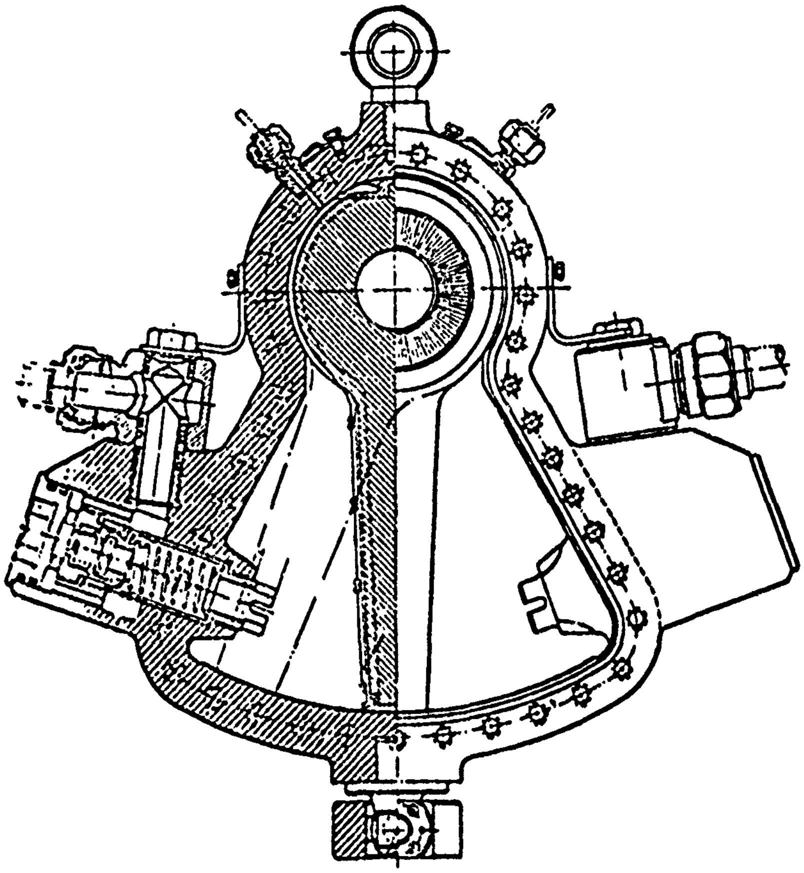

Figure 3-3. Vane type hydraulic actuator (general assembly)

An electrical signal, which may be as small as 1 ma, may or may not require amplification depending upon the torque motor or voice coil used to position a slapper. Position of the nozzle, or other throttling device, controls the first stage pressure relationship which positions the spool valve of the second stage. This stage creates a flow or pressure gain to position the piston of the actuator. Mechanical or electrical feedback relative to piston rod position creates a force balance with a feedback spring or a circuit balance respectively in a summing amplifier.

SECTION 4
ACTUATOR CLASSIFICATION

Actuators have been classified in accordance with the mechanism and type. Figure 4-1 shows the coding or numbering system used for actuators. Further classification may be based on individual characteristics.

Other than whether they were hydraulically or pneumatically operated, the available information did not specify further details on the actuators. Therefore, for analysis purposes the data was arranged into these two categories and failure rates were predicted accordingly.

Figure 4-1.

ACTUATOR CLASSIFICATION

PART NUMBER	ACTUATOR NAME
27 00 00	Actuator
27 01 00	Hydraulic Actuator
27 01 01	Linear (Hydraulic)
27 01 02	Rotary
27 01 03	Plunger
27 01 04	Vane
27 02 00	Pneumatic Actuator
27 02 01	Spring and Diaphragm
27 02 02	Springless Diaphragm
27 03 00	Motor Operated
27 03 01	Electric Motor
27 03 02	Electropneumatic
27 03 03	Electrohydraulic
27 04 00	Solenoid Actuator
27 04 01	Pilot-operated

SECTION 5

FAILURE MECHANISMS AND MODES

Actuators of all types generally display similar failure characteristics. The primary failure mode for actuators in storage or in operation is excessive internal or external leakage. The failure mechanisms attributing to the leakage are primarily failed piston seals and aging of elastomeric seals.

Primary failure modes and attributing failure mechanisms for operation and storage of actuators are listed in Table 5-1.

Table 5-1. ACTUATOR FAILURE MODES/ MECHANISMS

FAILURE MODE	FAILURE MECHANISM
Internal Leakage	Piston Seals
External Leakage	Aging Elastomeric Seals
Sticking	Wear/Aging
Hysteresis	Friction
Pressure Drop	Unbalanced Force
Solenoid Failure	Short
Mechanical Binding	Contamination

5.1.1 Internal Leakage

As with a valve, the actuator has a leakage problem which is most serious particularly for long term stored actuators. This is a problem because a small leakage rate can deplete the supply of the flowing medium. The flowing medium may be corrosive or explosive and damage to equipment or personnel can result. Excessive internal leakage is attributed primarily to failed piston seals; however, contamination can also cause increased wear and leakage.

5.1.2 Hysteresis

The next failure mode, hysteresis, is a result of excessive friction between moving parts. Packing contributes to this effect because it must create a seal sufficient to hold line fluid within the body. Additional friction occurs in the guide, and very fine stem finishes are employed. Piston seal rings in cylinder actuators also offer resistance to movement and cause some hysteresis.

Other moving parts such as, the plug, diaphragm plate and stem are possible problems due to contamination and wear. After long periods of storage, sticking of sliding or contacting can be caused by 1) cold welding, 2) inadequate lubrication, 3) contamination, or 4) incorrect design.

5.1.3 External actuator leakage

External leakage is caused by leakage through or around seals. This is due to aging of elastomeric seals or static seals. To eliminate this failure mode, welded body construction is preferred and permanent connections such as brazed, welded, or swaged should be when the components are installed into the system.

SECTION 6

DATA COLLECTION

Data collection for actuators has required contacting numerous sources. The primary sources as mentioned in the summary are only a part of the data actually collected. However, it reflects the data analyzed in this report.

The primary data in this report represents experience on seven missile programs, three space programs and searches of current literature and reliability data centers.

Collection of data was accomplished via personal discussions with specialists throughout government and industry and via independent literature searches including computerized data. Several personal contacts have been established with knowledgeable personnel in several organizations from which data is sought.

SECTION 7

DATA ANALYSIS

Data received has been reviewed, separated and regrouped into data sets by actuator type, application, manufacturing process, quality control and field use.

In the data analysis the exponential distribution was considered applicable when a "better fit" of the data by other reasonable distributions could not be justified. Utilizing the exponential failure distribution, failure rates presented in this report were calculated both from data in which no failures were observed and from data in which failures were observed and recorded. The methods of calculation for both cases are presented below. As is customary in statistical estimation, methods of calculation of one-sided confidence interval estimates of failure rate are also presented.

Typically, the data in this report arose from documented results of different tests, storage intervals, and/or operational uses of n distinct specimens of the same component under essentially the same environmental conditions. The duration of such tests, storage intervals, and/or usages may or may not have been the same for all specimens.

Accordingly, denoting the time accumulated on the ith specimen by t_i^*, the total time, t, accumulated by the n specimens is calculated using

$$t = \sum_{i=1}^{n} t_i^*$$

Here, if the ith specimen failed during its period of observation, then t_i^* represents the time to failure; otherwise, t_i^* simply represents the total observed time (without failure). The total number of failing specimens is denoted by r.

Thus, all failure-rate estimates given in this report were calculated using

$$\hat{\lambda} = \frac{r}{\sum_{i=1}^{n} t_i^*} \qquad (r>0)$$

All failure-rate estimates cited above are known as "best" or statistical "point" estimates. However, a given point estimate is known to vary from sample to sample according to the underlying failure distribution of the specimens. Because of this inherent variation in the point estimate, it is customary to accompany the point estimate with an interval estimate and its confidence limits. The interval estimate specifies the range of probability values. The likelihood that the unknown failure rate, λ, is actually contained in the interval estimate is specified by the confidence limits.

The confidence intervals to be given in following reports are of the type $(0, \hat{\lambda}_C)$; that is, they state with confidence C that the unknown failure rate, λ, lies between zero and an upper confidence limit, $\hat{\lambda}_C$. Such confidence intervals are called "one-sided," since they effectively sta... ...th confidence C, that "the unknown failure rate is at most $\hat{\lambda}_C$."

Assuming that the distribution of failure times is exponential (that is, that it follows $\lambda e^{-\lambda t}$), the one-sided confidence limit $\hat{\lambda}_C$ is calculated using the formula:[1]

$$\hat{\lambda}_C = \frac{\chi^2_{(C;2r+2)}}{2t}$$

where $\chi^2_{(C;2r+2)}$ is the 100C percentile of the χ^2 distribution with 2r + 2 degrees of freedom.

The value of C used in this report is 0.9; that is, the 90 percent confidence limit, $\hat{\lambda}_{90}$, was calculated.

[1] Cf. B Epstein, "Estimation From Life Test Data," IRE Transactions on Reliability and Quality Control, No. RQC-9, April 1960.

7.1 Number of Failures Equal to Zero

This is a special case of the preceding subsection.

The total observation time, t, is calculated, as before. The point estimate of failure rate is always zero. This, in effect, is equivalent to stating that the MTBF is infinite. Since zero failure rates and infinite MTBF's are physically impossible, Epstein's [1] approach was adopted, and the point estimates given in this report for zero failures were calculated using

$$\hat{\lambda} = \frac{1}{\sum_{i=1}^{n} t_i^*}$$

It is clear that this method will usually result in a pessimistic estimate of a component's failure rate, because the method implies failure of one specimen at the termination of observation. Although this pessimism cannot be removed, it can be somewhat alleviated by calculation of a one-sided confidence interval. With such an interval, it can be stated at some level of confidence that the failure rate is no more than a given amount, where "no more than" implies that the rate actually may be lower.

The corresponding one-sided confidence limit, $\hat{\lambda}_C$, was calculated with $r = 0$; namely,

$$\hat{\lambda}_C = \frac{\chi^2_{(C;2)}}{2t}$$

As before, $\hat{\lambda}_{90}$ was calculated for this report.

[1] B. Epstein, Statistical Techniques in Life Testing, Technical Report No. 4, ONR Contract Nonr-2163(00), 15 January 1959, AD 211458.

SECTION 8

ACTUATOR STORAGE FAILURE RATES

The actuator storage data, Table 8-2, did not identify specific types other than whether they were used in a hydraulic or pneumatic system. Accordingly, storage failure rates were derived for these two categories as shown in Table 8-1.

TABLE 8-1. ACTUATOR STORAGE DATA SUMMARY

Type	Storage Hrs. x 10^6	Failures	λ_s (Fits)	90% one-sided confidence limit
Hydraulic	608.6	121	(9.8)199(40769)	269
Pneumatic	239.0	21	(63) 88(256)	118

The numbers in parenthesis indicate the range of failure rate computed from individual data sources showing at least one failure. The 90% one sided confidence limit is also shown.

8.1 Hydraulic Actuator Storage Data

Storage data on hydraulic accumulators consisted of over 608 million hours with 121 failures for an overall failure rate of 199.7 fits. Examination of Table 8-2 reveals a wide variation in failure rates among the individual sources. From sources showing at least one failure, the failure rate varies from a low of 9.8 fits to a high of 40,769 fits. In an attempt to determine the reasons for this variance, the individual sources were reviewed for clues. Although the information on actuator types, storage environment, quality grades, length of storage and types of failures was not sufficient to reach absolute conclusions, several pcssibilities for the variance were identified.

a) Periodic exercising - Some of the equipments in Table 8-2 were exercised periodically during storage (data samples 15 and 16). Others (data samples 4, 11, 12 and 14) were never exercised throughout the storage period. A third class of data did not specify whether or not the equipment had been exercised. The overall failure rate for hydraulic

TABLE 8-2. ACTUATOR STORAGE DATA

PART DESCRIPTION	FUNCTIONAL APPLICATION	ENV.	PART POP.	NO. OF FAILURES	PART HRS. x 10^6	FAILURE RATE x 10^{-6}	UPPER 90% CONFIDENCE	YR. OF REPORT
1. Hydraulic Actuator	Linear	GND	-	0	31.000	<0.030	.074	74
2. Hydraulic Actuator	Linear	SUB	-	5	6.012	0.832	1.554	74
3. Hydraulic	Air-to-Air MSL (Storage)	MSL GND	46	0	2.11	<.474	1.091	64
4. Hydraulic	Ballistic MSL (Storage)	MSL GND	90	43	1.5768	27.270	33.413	64
5. Hydraulic Actuator	Dormant	GND	-	-	-	0.50	-	69
6. Actuator	Hot Gas	GND	-	-	-	0.175	-	69
7. Actuator	-	MSL	-	29	440.2	.0659	.084	69
8. Actuator	Storage	-	-	-	-	0.07	-	65
9. Actuator	Electrohy-draulic (Storage)	MSL	-	-	-	.01	-	65
10. Hydraulic Actuator	-	-	147	4	18.028	0.221	.444	63
11. Hydraulic Actuator	Desert Storage	-	5	0	0.7446	<1.343	3.102	63
12. Hydraulic Actuator	Aircraft	-	7	5	0.12264	40.769	76.205	63
13. Hydraulic Actuator	Aircraft	AIR	-	-	0.4555	-	-	63
14. Hydraulic Actuator	Storage	-	108	27	1.419	19.027	24.641	63
15. Hydraulic Actuator	Storage	-	11	8	.9636	8.302	13.495	63
16. Hydraulic Actuator			117	0	3.9288	<0.254	.588	63

TABLE 8-2. ACTUATOR STORAGE DATA

PART DESCRIPTION	FUNCTIONAL APPLICATION	ENV.	PART POP.	NO. OF FAILURES	PART HRS. x 10^6	FAILURE RATE x 10^{-6}	UPPER 90% CONFIDENCE	YR. OF REPORT
17. Hydraulic Actuator	Piston	-	6992	0	102.003	$<.0098$	.022	71
18. Actuator	Linear	GND	-	0	0.628	<1.592	3.677	74
19. General Actuator	Explosive	GND	-	13	207.100	0.063	0.100	74
20. Explosive	Storage Explosive	GND	-	8	31.260	0.255918	0.415	66

actuators is 199 fits. The failure rate of those equipments known to have not been exercised is almost 100 times the overall failure rate. If the data from non-exercised systems is removed from the overall failure rate computation, the ratio of non-exercised to the overall failure rate increases to over 250. Although it is difficult to ascertain the storage conditions of the third class of data, its overall failure rate is consistent with that of exercised systems. Based on this data, a trend toward higher failure rates for equipment which remains not exercised during storage is apparent and may partially account for the higher failure rates in Table 8-2.

b) <u>Changes in technology</u> - As time passes, advancements in technology are sometimes reflected in improved failure rates. For example, welded body construction and improved elastomeric seals result in improved reliability. The data sources in Table 8-2 vary in age from 1963 to 1974. The lowest failure rates are obtained from data samples 1 and 7 taken from reports dated 1974 and 1969 respectively. Those two sources, by virtue of their large number of hours (over 75% of all the hours), bias the overall failure rate toward the low end. All of the high failure rate items were taken from reports dated 1963 and 1964. Therefore, it is probable that a decrease of failure rate with time (and improved technology) is shown by the data.

c) <u>Complexity</u> - The complexity of an actuator can vary from a simple linear piston type to a complex reciprocating vane type. Since the data did not call out specific types, it is possible that the variance is due to differing degrees of complexity.

d) <u>Length of storage</u> - The length of storage for individual sources varied from a few years to as many as 17 years. Some of the failures were attributed to aging of seals. In this case, the failure rate of equipments stored for longer times would tend to be higher. In spite of this,

an exponential failure rate was still assumed since the data was not conclusive enough to establish an aging mechanism.

e) Reporting methods - Failures were counted on the basis of failure statements on failure reports. With the exception of the 6% of the reports concerned with corrosion, the balance of the failures may not have been caused by the storage environment.

8.2 Pneumatic Actuator Storage Data

Storage data for pneumatic actuators consists of 238.988 million part hours with 21 failures for a failure rate of 87.9 fits. For programs reporting at least one failure, a range of failure rates from 9.8 fits to 256 fits was observed. Although this range is not as wide as that for hydraulic actuators, some of the reasons discussed in Section 8.1.1 may be responsible for the variation.

8.3 Operational/Non Operational Failure Rate Comparison

Operational to non operational failure rates ratios (K factors) were computed for hydraulic and pneumatic actuators. Operating data was taken from the RADC Nonelectronic Reliability Notebook. The data and K factors are shown below.

Type	λ_{op} (fits)	λ_s (fits)	λ_{op}/λ_s
Hydraulic	15288	199	77
Pneumatic	1507	88	17

SECTION 9

CONCLUSIONS AND RECOMMENDATIONS

9.1 Conclusions

In general, actuator types could not be identified except for the system type in which they were installed. Quality grades were not well defined for the actuator data collected. To determine quality grades extensive searching through component specifications and drawings would be required. Hence, effects of quality levels, if any, could not be determined.

There was no significant difference between dormant and storage reliability data. Dormant and storage data were combined in all analyses.

Storage data collected for each generic actuator type was not plentiful. Therefore, failure rates derived at this high level have higher statistical confidence than those of the sub-categories and should be utilized unless specific information is available to further define the type of actuator under consideration.

9.2 Recommendations

Storage failure rates for hydraulic and pneumatic actuators are as follows:

Type	λ(fits)
Hydraulic	199
Pneumatic	88

Although a wide variance was observed among the different sources, the above failure rate is recommended for prediction. This failure rate is representative of missile actuators stored under varying conditions and of different quality grades.

Record keeping for actuators kept on storage should be improved, specifically the identification of quality grades and actuator description. This should be done within existing data collection systems.

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only by their generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented for all missile hydraulic and pneumatic systems.

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This report documents findings on the non-operating reliability of actuators. The study consists of over 864 million part hours. The sources show an overall storage failure rate of 199 FITS(failures per billion hours) with a one sided 90% confidence limit of 269 FITS for hydraulic actuators. For pneumatic actuators the storage failure rate is 38 FITS and the one sided 90% confidence limit is 118 FITS. Elements of actuator design are reviewed. This information is part of a research program being conducted by the

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20. Abstract (continued)

U. S. Army Missile Command, Redstone Arsenal, Alabama. The objective of this program is the development of non-operating (storage) reliability prediction and assurance techniques for missile materiel.